Applied and Computational Mechanics 5 (2011) 187–196

Applied and Computational Mechanics

Reduction of lateral forces between the railway vehicle and the track in small-radius curves by means of active elements

T. Michálek[a,*], J. Zelenka[a]

[a] *University of Pardubice, Jan Perner Transport Faculty, Detached Branch Česká Třebová, Slovanská 452, 560 02 Česká Třebová, Czech Republic*

Received 4 October 2011; received in revised form 15 December 2011

Abstract

This paper deals with a possibility of reduction of guiding forces magnitude in small-radius curves by means of active elements. These guiding forces characterize the lateral force interaction between the rail vehicle and the track and influence the wear of wheels and rails in curves. Their magnitudes are assessed in the framework of vehicle authorization process. However, in case of new railway vehicles with axleload of approximately 20 t and more it is problematic to meet the condition of maximum value of the quasistatic guiding force which acts on the outer wheel of the 1st wheelset in small-radius curves. One of the possible ways how to reduce these forces is using the system of active yaw dampers. By means of computer simulations of guiding behaviour of a new electric locomotive, comparison of reached values of the quasistatic guiding forces in case of locomotive equipped with active yaw dampers and without them was performed. Influences of magnitude of force generated by the active yaw dampers, friction coefficient in wheel/rail contact and curve radius were analysed in this work, as well.

Keywords: active yaw dampers, quasistatic guiding forces, electric locomotive, simulations

1. Introduction

Under the term “guiding behaviour” we understand dynamical properties of railway vehicles during the passing through curves. The guiding behaviour is assessed in the framework of authorization process of new or modernised vehicles according to the European Standard EN 14363 [2] or the UIC Code No. 518 [6]. So-called quasistatic guiding force is one of the most important quantities in this respect. Guiding forces characterize the lateral force interaction between the vehicle and the track; if the rail vehicle passes through a curve these forces act on each wheel in wheel/rail contact and influence the wear of wheels and rails.

In case of modern locomotives with axleload of approximately 20 t and more (especially if they are designed for high speed operation or for freight haulage) it is problematic to meet the condition that value of the quasistatic guiding force acting on the outer wheel of the 1^{st} wheelset in small-radius curves should not exceed the limit value $Y_{qst,lim} = 60$ kN [2]. In case of exceeding of this value the authorization process of the vehicle is complicated; due to higher degree of wear of wheels and rails, operational costs of such locomotive are higher, as well. Therefore, producers of railway vehicles search for technical solutions which allow reduction of the quasistatic guiding forces during the passing through curves. For a long time, bogie couplings have been used for these purposes in case of electric locomotives. However, an ability of the bogie couplings to reduce the guiding forces is limited. A more effective solution how to reduce the magnitudes of guiding forces is represented by active elements.

*Corresponding author. Tel.: +420 465 533 006, e-mail: tomas.michalek@student.upce.cz.

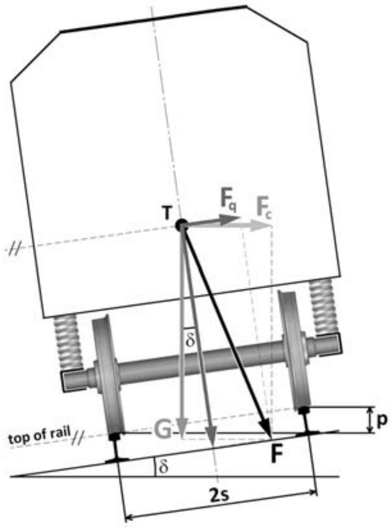

Fig. 1. Derivation of quantities F_q and a_q

Such a system of active elements, which serves for reduction of lateral forces in wheel/rail contact during the passing through curves (and wear of wheels and rails as well as operational costs, consequently), was developed by Liebherr and operationally tested on several Siemens' electric locomotives in 2006 [1]. In 2008 Czech producer of rail vehicles Škoda Transportation started testing of its new electric locomotive type 109E. Since 2009 an influence of hydraulic bogie coupling on the guiding behaviour of this locomotive has been observed, as well; see [9]. The aim of this paper is to assess the possible influence of system of active elements on guiding behaviour in small-radius curves if this device would be implemented into the running gear of the locomotive Škoda 109E.

2. Technical solutions for reduction of quasistatic guiding forces

If a standard four-axled locomotive with two bogies runs through a curve the maximum value of quasistatic guiding force usually acts on the outer wheel of the 1^{st} wheelset and the second highest value of this force belongs to the outer wheel of the 3^{rd} wheelset. Quasistatic guiding forces act also on all other wheels but their magnitudes are usually not so high. However, the sum of all these forces must always correspond to the Newton's law, i.e.:

$$\sum Y_{qst} = M_{loc} \cdot a_q = F_q, \tag{1}$$

where M_{loc} is total weight of locomotive and a_q is so-called unbalanced lateral acceleration. The product of M_{loc} and a_q is also named as unbalanced centrifugal force F_q.

Quantity a_q is often used in the branch of railway vehicle dynamics (see for example [6] or [4]) and its value can be derived on the base of Fig. 1. Unbalanced centrifugal force F_q — i.e. the component of resultant force F which is parallel with plane of the top of rail — is given as:

$$F_q = F_c \cdot \cos\delta - G \cdot \sin\delta, \tag{2}$$

where F_c is centrifugal force, G is force of gravity and δ is angle of the cant. This angle is given by superelevation p and tape line distance $2s$ (in case of standard-gauge rail vehicles, the tape line distance has a value of $2s = 1\,500$ mm) and its value is relatively small. Therefore, the force F_q can be approximately expressed as:

$$F_q = M_{loc} \cdot \frac{v^2}{R} - M_{loc} \cdot g \cdot \frac{p}{2s}, \tag{3}$$

where v is vehicle speed, R is curve radius and $g = 9.81$ m $\cdot$ s^{-2} is acceleration of gravity. It is evident that the unbalanced lateral acceleration a_q is given as:

$$a_q = \frac{v^2}{R} - g \cdot \frac{p}{2s}. \tag{4}$$

It is evident that under concrete conditions, which are defined by values of weight M_{loc} and unbalanced lateral acceleration a_q, the sum of all quasistatic guiding forces ΣY_{qst} must have an always constant value given by equation (1). A very important consequence of this equation is the fact that each technical device for reduction of quasistatic guiding forces can only redistribute these forces among the other wheels. It means that the maximum (and from the point of view of the current EN standard [2] also critical) values can be decreased; however, the sum of all these forces must stay unchanged.

2.1. Devices for implementation into the primary suspension

Practically all technical devices for reduction of quasistatic guiding forces work in such a way that they decrease the value of angle of attack of relevant wheelset or bogie. The angle of attack of the wheelset can be influenced by means of primary suspension and wheelset guiding. *Passive system of radial wheelset steering*, based on low stiffness of the wheelset guiding in longitudinal direction (possibly supplemented with cross-coupling of wheelsets), is usually not suitable for high speed vehicles because of problems with stability of run at higher speeds.

Nowadays, *active systems of radial wheelset steering* are being developed. These systems usually work in such a way that axleboxes of each wheelset are linked with bogie frame by means of actuators which allow steering of the wheelset into the radial position, i.e. decreasing of the angle of attack during the passing through curves. A technical device using this principle is being developed for example by Bombardier as an equipment of its “Flexx Tronic” bogie for double-decker EMUs — see [5, 11]; it is named ARS (in German: “aktive Radsatzsteuerung”). The principle of active wheelset steering generally allows improving the stability of run in straight track (it means increasing of the critical speed of the vehicle), as well. However, these active systems are not used in technical practice in the present.

2.2. Devices for implementation into the secondary suspension

Technical solutions for reduction of quasistatic guiding forces, which belong into this category, influence angle of attack of the whole bogies. These systems can be designed passive as well as active. *Bogie coupling* is one of the widely used passive systems. The bogie coupling turns the bogies of the vehicle during the passing through curves about the vertical axis into opposite directions. The best state, which can be reached in this way, is equalisation of the quasistatic guiding forces acting on outer wheels of the 1$^{\text{st}}$ and 3$^{\text{rd}}$ wheelset. The bogie coupling (the force transmission between the bogies) can be realized mechanically as well as hydraulically. Mechanical bogie coupling is often applied in running gears of older electric locomotives. As it was mentioned, hydraulic bogie coupling was tested for example on the locomotive Škoda 109E; a more detailed description of this device as well as results of computer simulations are shown in [9].

Usage of active elements, especially *active yaw dampers*, is the second possibility how to reduce quasistatic guiding forces. During the passing through curve, this device works in such

Fig. 2. Active yaw damper (ADD) from Liebherr [11]

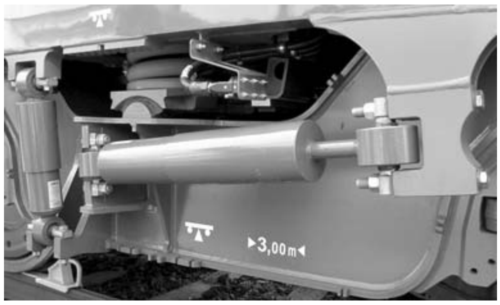

Fig. 3. Preparation for installation of ADD on the locomotive Siemens Vectron

a way that both bogies of the vehicle are turned into the bogie direction; it means that angles of attack of both bogies are decreasing. In straight track, these dampers work as classic yaw dampers; it means that they damp yawing motion of bogies. System of active yaw dampers, which is known as ADD (in German: "aktive Drehdämpfer"), was developed in co-operation of Siemens and Liebherr and operationally tested on 6 Siemens' electric locomotives (types ES64F4 and ES64U2); see for example [1, 11]. An advantage of this system is the fact that actuator is integrated into the yaw damper (see Fig. 2) and its installation does not need any modifications of the bogie or vehicle body frame. ADD seems to be a very effective way how to reduce the magnitudes of quasistatic guiding forces in small-radius curves and Siemens want to use this system on its new generation of locomotives "Vectron", as well. Preparation for installation of ADD instead of classic yaw dampers (i.e. installation of the power supply connectors) on the electric locomotive "Vectron" is shown in Fig. 3.

3. Simulations of guiding behaviour of an electric locomotive

For purposes of assessment of the active yaw dampers influence on guiding behaviour of an electric locomotive series of computer simulations was performed. The simulations were carried out by means of multi-body program system "SJKV" which is being developed at the Detached Branch of the Jan Perner Transport Faculty in Česká Třebová. Dynamical model of the electric locomotive Škoda 109E was used and modified for these purposes. Magnitudes of the quasistatic guiding forces reached with locomotive equipped with active yaw dampers were compared with results of locomotive without them.

3.1. Program system "SJKV" for simulations of rail vehicle dynamics

The program system "SJKV" is original multi-body simulation (MBS) software for simulations of running and guiding behaviour of rail vehicles which is being developed in the IDE Borland Delphi. Its architecture is based on program units and allows creating of different modifications for concrete rail vehicles. In comparison with commercial MBS software as for example SIMPACK or MSC ADAMS, the greatest advantage of the system "SJKV" is a detailed knowledge of algorithms on which the calculations are based. In case of the commercial MBS software, "black-box-principle" from the user's point of view is often a source of problems, especially at the verification of computational models.

In the program system "SJKV", the whole non-linear system vehicle–track is modelled as a multi-body system; it means that all the bodies are considered rigid and coupled by means of elastic and damping couplings. This dynamical model of the system vehicle–track is mathematized on the base of *structural elements method*; therefore, one equation of motion belongs to each degree of freedom of the multi-body system. So, the acceleration vector is generally given by equation:

$$\ddot{\mathbf{q}} = \mathbf{M}^{-1} \cdot \mathbf{L} \cdot \mathbf{F}, \tag{5}$$

where $\mathbf{M}$ is mass matrix, $\mathbf{L}$ is geometric matrix and $\mathbf{F}$ is vector of acting forces. These forces include forces of gravity, wheel/rail contact forces and coupling forces. In general case, characteristics of these couplings are non-linear. Components of the acceleration vector represent the acceleration of relevant bodies in directions of considered degrees of freedom (i.e. translation and rotation of these bodies).

Solving of mathematical model of the system is based on *finite differences method*. In this way, deflections and velocities in following time integration step can be calculated by means of deflections and accelerations in two previous steps; precondition for using of this method is a constant value of the acceleration during the time integration step Δt. So, the deflection vector and the velocity vector in the time step $i+1$ are given as:

$$\mathbf{q}_{i+1} = 2 \cdot \mathbf{q}_i - \mathbf{q}_{i-1} + \ddot{\mathbf{q}}_i \cdot (\Delta t)^2, \tag{6}$$

$$\dot{\mathbf{q}}_{i+1} = \frac{1}{2 \cdot \Delta t} \cdot (3 \cdot \mathbf{q}_{i+1} - 4 \cdot \mathbf{q}_i + \mathbf{q}_{i-1}). \tag{7}$$

Special attention is paid to solving of wheel/rail contact in the program system “SJKV”. Forces acting in the wheel/rail contact can be divided into 3 categories — wheel forces Q acting in vertical direction, guiding forces Y acting in lateral direction and longitudinal creep forces T_x. The lateral and longitudinal forces Y and T_x, which intermediate the adhesion joint between wheels and rails, are computed by means of algorithm proposed by Polách [4]. For purposes of using in the simulations, the wheel/rail contact is described by means of *characteristics of the wheel/rail contact geometry*. These characteristics are computed on the basis of wheel and rail profiles, track gauge, rail inclination and wheelset gauge by means of original program system “KONTAKT 5” and include:

- *position of the contact points on wheels and rails* in dependency on lateral movement of the wheelset y_w,

- *delta-r function* which gives actual roll radius difference of the left and right wheel of the wheelset in dependency on lateral wheelset movement y_w; it is given as:

$$\Delta r = r_L - r_R = f(y_w), \tag{8}$$

- *tangent-γ function* which is given by rake angles of contact planes on the left and right wheel in dependency on lateral wheelset movement y_w and which is an important parameter for the force interaction between the wheelset and the track; it is given as:

$$\tan\gamma = \tan\gamma_L - \tan\gamma_R = f(y_w), \tag{9}$$

- *equivalent conicity* which characterize intensity of wheelset centring effect. This quantity depends on amplitude of lateral wheelset motion y_0. Value of the equivalent conicity for the wheelset amplitude of $y_0 = 3$ mm is often used as comparative value of various wheelset/track contact pairs; this method of assessment of the wheel/rail contact geometry during the testing of running and guiding behaviour of railway vehicles is defined in standards [2] and [6], as well.

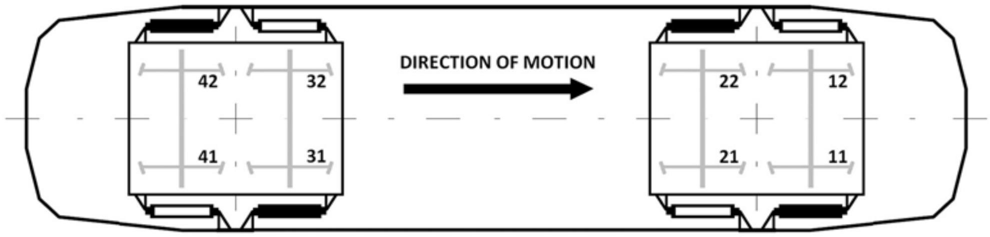

Fig. 4. Considered arrangement of classic (white) and active (black) yaw dampers on the locomotive

As it was said, mathematical description of the dynamical model is the basis of the MBS system "SJKV". Therefore, *input data* for simulations include above all dimensional and mass parameters of the model, characteristics of couplings and characteristics of contact geometry. Next important input data are given by alignment of the track, track irregularities and vehicle speed. More detailed description of the system "SJKV" is presented in [7].

3.2. Modification of the dynamical model of the locomotive

For purposes of assessment of active yaw dampers influence on guiding behaviour, dynamical model of the locomotive Škoda 109E had to be modified. Therefore, modelling of the system of active yaw dampers was the most significant modification in this case. New version of the program system named "SJKV-L3A" was created in this way.

Dynamical model of the vehicle consists of 7 rigid bodies, i.e. 4 wheelsets, 2 bogies and vehicle body. Smaller bodies (as for example springs, dampers or components of traction drive) are reduced to the considered bodies. This degree of simplification of physical reality seems to be fully sufficient because results of simulations performed with using of the system "SJKV" (see for example [9]) show relatively good agreement with results of measurements. Dynamical model of the track is created with reduced rail mass which belongs to each wheel (see [7]); it comprises 8 additional rigid bodies. In this case, the whole system vehicle–track has 50 degrees of freedom. Couplings between the bodies (i.e. springs, wheelset guiding, rotational stiffness bogie–vehicle body etc.) are considered elastic and damping; coupling forces are computed on the basis of deflections and velocities of the rigid bodies, i.e. deformations of the couplings and velocities of these deformations. Characteristics of considered couplings are usually given by producer of the vehicle or measured on test stand (see for example [10]).

Locomotive Škoda 109E is standardly equipped with 8 yaw dampers, i.e. 4 yaw dampers per bogie. In the framework of this work, 4 classic yaw dampers (i.e. 2 pieces per bogie) were replaced with active dampers in dynamical model. System of the active yaw dampers, which is technically based on the system ADD, was considered. It means that these dampers generate constant (pull or pressure) force, if the vehicle runs through small-radius curves, and help so with the radial steering of bogies. Small-radius curves are identified by means of angles between the bogies and the vehicle body, using the displacement sensors integrated into the active dampers. In Fig. 4 there is shown considered arrangement of classic and active dampers on the locomotive.

Arrangement of the active yaw dampers is designed in such a way that all these dampers generate either pull or pressure force at the same time. During the passing through curve, the active yaw dampers on the front bogie act in the opposite direction in comparison with moment against the bogie rotation; on the rear bogie, moment of the active dampers forces and the moment against the bogie rotation act in the same direction. So, total moment about vertical axis between the first/second bogie and the vehicle body is given as:

$$M_{I/II} = F_{act} \cdot 2W_{st} \mp \gamma \cdot \beta_{I/II}, \tag{10}$$

where F_{act} is force generated by the active yaw damper, $2W_{st}$ is lateral distance between yaw dampers, γ is rotational stiffness of secondary suspension and β is angle about vertical axis between the relevant bogie and the vehicle body during the passing through curve. In consequence of active elements actuation, additional moment about vertical axis acts on vehicle body, as well. Therefore, its magnitude must be compensated by means of additional lateral forces in secondary suspension — i.e. by deflection of flexi-coil springs, possibly by lateral forces acting on bogie bump stops.

3.3. Simulations of the locomotive equipped with the system of active yaw dampers

Simulations of guiding behaviour of the locomotive were performed for passing through small-radius curves (so-called "testing area No. 4" according to [2]) which is usually critical from the point of view of the limit value of quasistatic guiding force. Concretely, curves with radius of $R = 250$ m, 300 m and 350 m were investigated; superelevation in all the curves had value of $p = 150$ mm. For purposes of this work, only quasistatic calculations (i.e. simulations on ideal track without irregularities) were performed. However, this simplification should not have any significant influence on accuracy of results. As it was proved (see for example [8]), in case of assessment of the quasistatic guiding forces, results of the quasistatic calculations differ less than 1 % from mean values obtained by means of time-consuming statistic processing of results of dynamic calculations (i.e. simulations on real track with measured irregularities). The vehicle speed was chosen in such a way that it corresponded to unbalanced lateral acceleration of $a_q = 1.1$ m $\cdot$ s^{-2} — it means: $V = 81.7$ km/h in the *R*250-curve, $V = 89.5$ km/h in *R*300-curve and $V = 96.7$ km/h in *R*350-curve. This value of unbalanced lateral acceleration follows from requirements of the standards [2] and [6] on maximum testing vehicle speed in small-radius curves.

Conditions of wheel/rail contact geometry were given by measured (slightly worn) wheel profile S1002 and theoretical rail profile 60E1 with inclination $1 : 20$; this contact pair is characterized with low value of equivalent conicity. Value of the friction coefficient in wheel/rail contact was changed in the range of 0.2 up to 0.5. For purposes of assessment of active yaw dampers influence on quasistatic guiding forces magnitudes, value of the force F_{act} generated by active elements was changed in the range of 5 up to 20 kN, as well.

4. Simulation results

As it was said in previous chapter, influence of the force generated by active yaw dampers as well as influences of the friction coefficient in wheel/rail contact and the curve radius on the guiding behaviour of the locomotive Škoda 109E were observed and subsequently analysed. Simulation results reached with locomotive equipped with the system of active yaw dampers were compared with quasistatic guiding forces of locomotive without them.

In bar chart in Fig. 5 there are shown magnitudes of the quasistatic guiding forces reached on outer wheels of the 1$^{\text{st}}$ and 3$^{\text{rd}}$ wheelset (i.e. guiding wheels in both bogies) in a curve with radius $R = 300$ m at the speed $V = 89.5$ km/h (i.e. $a_q = 1.1$ m $\cdot$ s^{-2}) for various values of friction coefficient in wheel/rail contact. It is evident that the system of active yaw dampers has a significantly positive influence on quasistatic guiding forces acting on guiding wheels. In comparison with the 1$^{\text{st}}$ wheelset, decreasing of the quasistatic guiding force magnitude on the

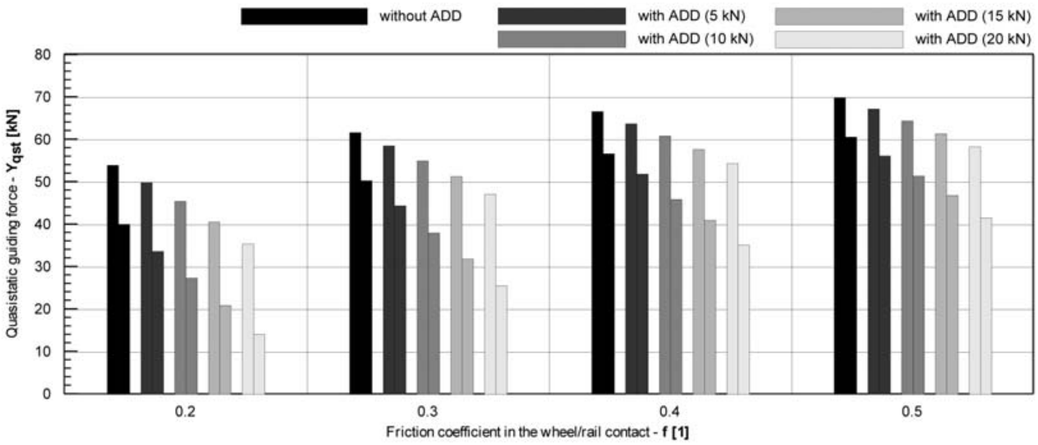

Fig. 5. Quasistatic guiding forces acting on outer wheels of the 1st and 3rd wheelset in curve with radius $R = 300$ m for various values of friction coefficient in wheel/rail contact and for various setting-up of maximum force generated by active yaw dampers

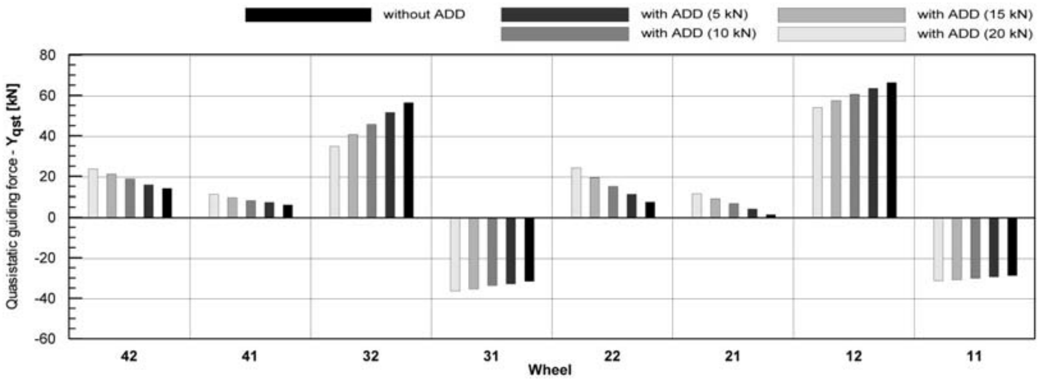

Fig. 6. Quasistatic guiding forces acting on constituent wheels in curve with radius $R = 300$ m for value of friction coefficient in wheel/rail contact $f = 0.40$ and for various setting-up of maximum force generated by active yaw dampers

3rd wheelset in dependency on increasing force generated by active elements is more significant. This fact can be explained with positive effect of the moment against bogie rotation in case of the rear (second) bogie — see equation (10). In case of normal friction conditions (it means that value of friction coefficient in wheel/rail contact belongs into the range of 0.3 up to 0.4), value of the force generated by active elements of $F_{act} = 10$ kN seems to be sufficient to meet the condition of the limit value of quasistatic guiding force $Y_{qst,lim} = 60$ kN which should not be exceeded in curves with mean value of radius $R_m = 300$ m ($+50$ m/-20 m) according to [2].

Influence of active yaw dampers on distribution of quasistatic guiding forces on constituent wheels of locomotive is shown for value of friction coefficient $f = 0.40$ and curve radius $R = 300$ m in bar chart in Fig. 6. It is evident that equation (1) must be always valid — in consequence of ADD actuation, magnitudes of guiding forces acting on wheels No. 12 and 32 (guiding wheels in both bogies) decrease; however, magnitudes of guiding forces acting on wheels of the 2^{nd} and 4^{th} wheelset increase. Numbering of wheels, which is used in bar charts in Fig. 6 up to 8, is explained in Fig. 4; the first number marks number of wheelset, the second number marks number of wheel of this wheelset. Considered motion of the locomotive has direction from the left to the right (see Fig. 4); so, the guiding wheels in the bogies in right hand curve are the wheels No. 12 and 32.

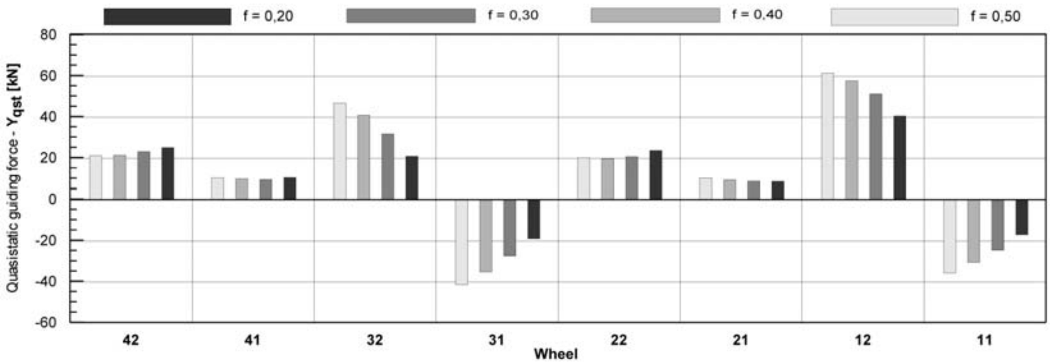

Fig. 7. Quasistatic guiding forces acting on constituent wheels in curve with radius $R = 300$ m for value of force generated by active yaw dampers $F_{act} = 15$ kN and for various values of friction coefficient in wheel/rail contact

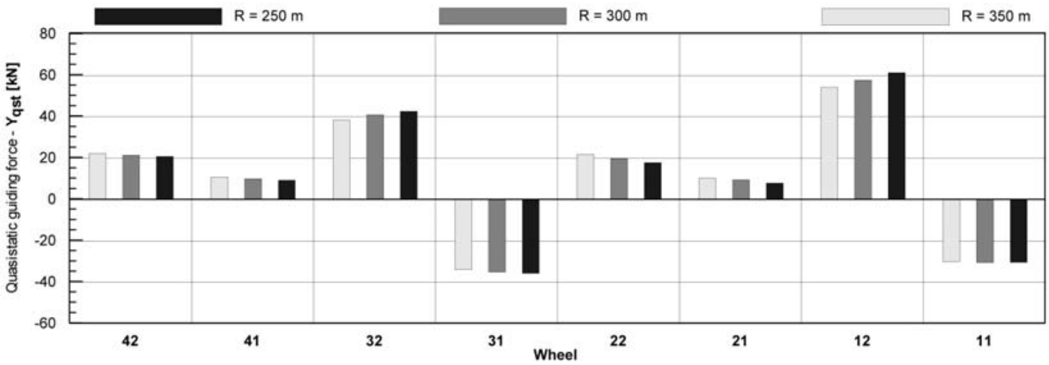

Fig. 8. Quasistatic guiding forces acting on constituent wheels in various curves for value of force generated by active yaw dampers $F_{act} = 15$ kN and for friction coefficient $f = 0.40$

Bar chart in Fig. 5 also confirms the fact that value of friction coefficient in wheel/rail contact has a very significant influence on magnitudes of quasistatic guiding forces; see for example [3], as well. Influence of friction coefficient on distribution of these forces on constituent wheels is demonstrated in Fig. 7. It is apparent that friction coefficient influences especially the forces acting on wheels of guiding wheelsets (i.e. the 1^{st} and 3^{rd} wheelset); equilibrium according to equation (1) must be retained again.

Influence of curve radius on quasistatic guiding forces was observed, as well. This influence is demonstrated for concrete conditions (given by values of friction coefficient and force generated by active elements) in bar chart in Fig. 8. In all cases the vehicle speed corresponded to unbalanced lateral acceleration $a_q = 1.1\ \mathrm{m \cdot s^{-2}}$. In comparison with influences of active yaw dampers and friction coefficient, this effect is relatively small.

5. Conclusion

This paper deals with a possibility of reduction of quasistatic guiding forces during the passing through curves by means of system of active yaw dampers. Except the analysis of the influence of these active elements, effects of friction coefficient in wheel/rail contact and curve radius were observed by means of computer simulations.

On the basis of reached results it is possible to say that the system of active yaw dampers seems to be an effective way how to reduce magnitudes of quasistatic guiding forces and consequently wear of rails in curves and wheels, as well. Lower degree of wear leads to lower operational costs, too; this effect, which is very important for carriers, is more significant if the fee for using of railway infrastructure depends on the effects of the vehicle on the track. In spite of this, using of active yaw dampers is not usual nowadays. Implementation of active elements into the running gear of rail vehicle puts very high requirements on safety; therefore, authorisation process of these systems is very complicated and expensive.

However, performed simulations show that implementation of the system of active yaw dampers (ADD) into secondary suspension of the locomotive Škoda 109E can cause more significant improvement of its guiding behaviour than the hydraulic bogie coupling which was tested on this vehicle recently (see [9]). For purposes of meeting the conditions of the European Standard EN 14363 [2] on the maximum value of quasistatic guiding force in the area of small-radius curves (so-called "testing area No. 4"), the system ADD, which is able to generate the maximum force $F_{act} = 10$ up to 15 kN, seems to be sufficient. It is important to say that many other conditions (for example the friction coefficient in wheel/rail contact or moment against the bogie rotation) have substantial influence on vehicle guiding behaviour, as well. However, application of active elements into the running gear of locomotives could represent the way how to design more effective railway vehicles in the future.

Acknowledgements

This work was supported by internal grant project of the University of Pardubice No. 51030/20/ SG510001 and by project of the Ministry of Education, Youth and Sports No. 1M0519 — Research Centre of Rail Vehicles.

References

[1] Breuer, W., Active torsional damper — an innovative damper concept in operational use, (in German), Eisenbahntechnische Rundschau 4 (2007), p. 186–189.

[2] ČSN EN 14363:2006. Railway applications — Testing for the acceptance of running characteristics of railway vehicles — Testing of running behaviour and stationary tests, (in Czech), Czech Institute for Normalisation, 2006.

[3] Michálek, T., Analysis of the influence of the friction coefficient on the guiding behaviour, Conference proceedings of the 13th conference Applied Mechanics (2011), p. 147–150.

[4] Polách, O., A Fast Wheel-Rail Forces Calculation Computer Code, Proceedings of the 16th IAVSD Symposium,Vehicle System Dynamics Supplement 33 (1999), p. 728–739.

[5] Schneider, R., Swaying-compensation and active wheelset steering "Flexx Tronic" from Bombardier, (in German), Eisenbahn-Revue International 4 (2010), p. 174–181.

[6] UIC Code 518:2009. Testing and approval of railway vehicles from the point of view of their dynamic behaviour — Safety — Track fatigue — Running behaviour, 4th edition, International Union of Railways (UIC), 2009.

[7] Zelenka, J., Running and guiding behaviour of two-axled diesel-electric locomotives CZ LOKO, (in Czech), New Railway Technique 6 (2009), p. 15–23.

[8] Zelenka, J., Michálek, T., A New Method of the Assessment of Rail Vehicles Guiding Behaviour in Small-radius Curves, International Journal of Applied Mechanics and Engineering 15 (2) (2010) 511–519.

[9] Zelenka, J., Špalek, P., Simulations of the guiding behaviour of the locomotive Class 380, assessment of the bogie coupling influence, (in Czech), Proceedings of the 19th conference with international participation Current Problems in Rail Vehicles (2009), p. 195–198.

[10] Zelenka, J., Vágner, J., Hába, A., Experimental verification of possibilities of determination of flexi-coil springs lateral stiffness, (in Czech), Vědeckotechnický sborník ČD 31 (2011).

[11] Liebherr — InnoTrans 2010, Berlin, promotional material.